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METHODOLOGY OF COMPUTING ELECTRIC ENERGY CONSUMPTION BY THERMAL PUMP SYSTEM AND SEASONAL EFFICIENCY

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ABSTRACT

The present article has studied the thermal pimp system of air-water type, maintaining heat production for the building in Almaty (Kazakhstan). The thermal pump system's working mode in winter time is considered as intermittent. Thermal pump system's heat characteristics, environment external temperature values have been conducted using the real data. The methodology for calculating the energy consumption and seasonal efficiency of the heat pump was calculated. The minimum temperature for heating the heat pump is determined, and the duration of operation at the temperature of the outside air is also calculated.

KEYWORDS: Solar Collector, Heat Pump, Seasonal Efficiency & Thermosyphon Circulation

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Original Article

1 INTRODUCTION

Upon buildings heating, there grows the need in ecological solutions. When heating buildings, the need for environmental solutions increases. An energy audit of the educational buildings of the University of Food Technology (UFT) was conducted in Plovdiv, Bulgaria [1-2]. It makes them effective both for the known ecological problems, linked with direct and indirect carbon dioxide emission into atmosphere and for heating and cooling. Moreover, they satisfy consumers' economic efficiency in respect of quality [3-4]. Thermal pumps seasonal characteristics upgrade and ever growing energy consumption by buildings have aroused much interest [5-6]. The ambient temperature, which has a great influence on the characteristics of the heat pump, is also important. The main point in the study of the energy efficiency of buildings is the operation of the heat source and the determination of its operational characteristics throughout the heating season.

Connolly and coauthors [7] have submitted a complex scenario, in which the thermal pumps play the key role in the renewable energy systems. Mathiesen et al. [8] distinguish, that thermal pumps increase the energy systems flexibility. Thermal pumps performance raising has special significance, as energy consumption during operation causes the biggest overall emission in carbon dioxide equivalent [9]. The work [10] shows, that correct modulation of the capacity for the heat needs considerably lowers the heating pump consumption capacity, thus, compressor's operational speed considerably influences at thermal pump productivity [11]. Several researches have confirmed the compressor's speed effect at the output [9–12]. The work [13] considers studying the convective heat exchange in flay solar collectors. The dependences of the Nusselt criterion in round and flat pipes of the solar collector are obtained. In [14], the graph analytical method of the energy and optical characteristics of a flat solar collector is considered. The total solar radiation of the inclination of the solar collector is calculated. Optical characteristics coefficient has amounted to 66%. It confirms correctness of solar collector's parameters selection.

In the given work, we have done calculations, the aim of which is defining the thermal pimp heating system performance, as well, the methodology, which specifies the thermal pump seasonal work efficiency, used for administrative building heating in Almaty city (Kazakhstan). Experimental researches have been carried out in winter heating season. The thermal pump operation schedule for heating was from Monday to Friday from 6 am to 6 pm. An important thing for calculations accuracy is getting the accurate information about ambient temperature change in length of time. For ambient temperature, there has been used the data from Kazakhstan meteorological station for the heating season from c 15.10.2018 to 15.04.2019, Figure 1.

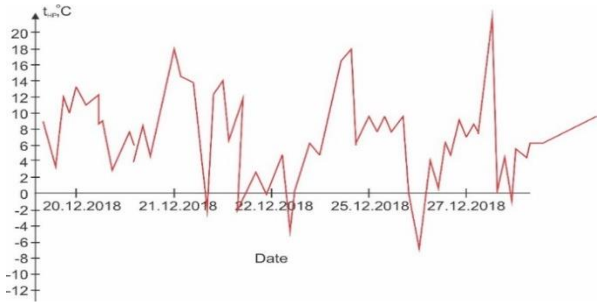


Figure 1: Outside Temperature Graph for the Period from 15.10.2018 to 15.04.2019.

Figure 2 shows outside temperature graphical dependence on hours amount during the heating season and hours amount, within which the thermal pump operates.

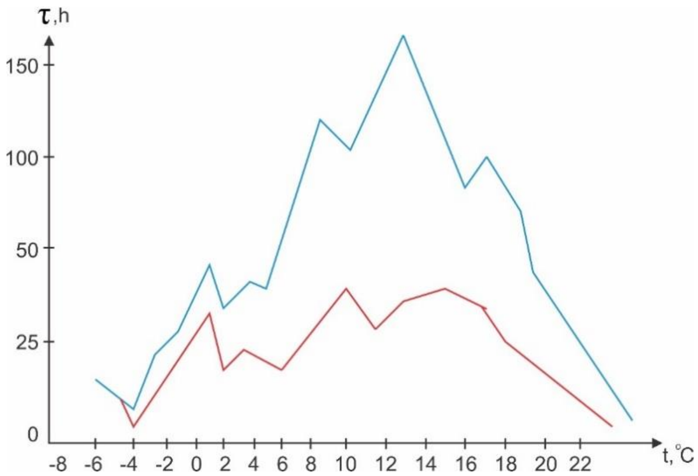


Figure 2: Outside Temperature Graphical Dependence on Hours amount, within which the Thermal Pump Functions.

2. METHODOLOGY OF COMPUTING ELECTRIC ENERGY, CONSUMED BY THERMAL PUMP SYSTEM AND SEASONAL OPERATION PERFORMANCE

Double circuit flat solar collector with thermosyphon circulation has been constructed and installed at the Institute of Information and computational technologies of MES RK, Almaty, Republic of Kazakhstan: 77 degrees of East longitude and 43 degrees of Northern latitude), as it is shown on the Figure 1. The installation has tubes over the absorber plate, which acts according to a thermosyphon principle (Figure 1 and 2). A flat collector slope angle is 45°, directed to the South. Water density change inside the reservoir creates thermosyphon activity.

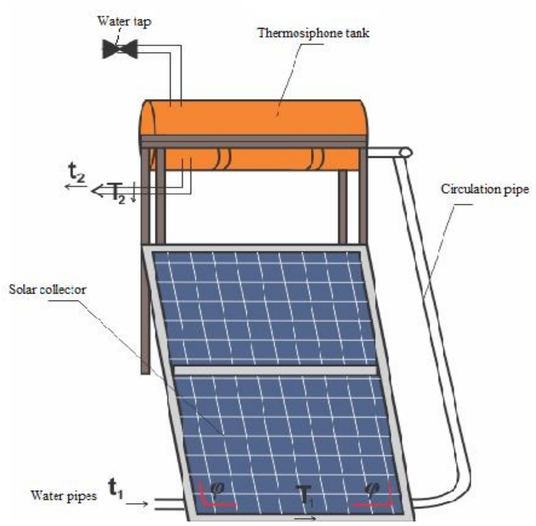


Figure 3: General View of Solar Water Heater.

Substantiation of design technological scheme: A dual-circuit solar installation with a heat pump has been developed (Figure 1), the installation is formed of four basic units: thermosyphon 1, solar collector 2, heat pump 3, tank accumulator 4.

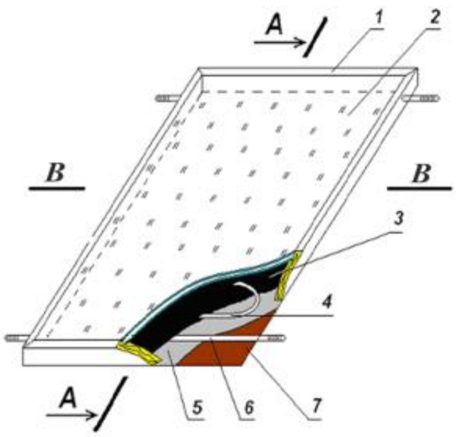


Figure 4: Flat Solar Collector.

Figure 2 demonstrates the flat solar collector model. The installation's principle and novelty consists in its difference from the known designing principle, the collector contains a transparent double glazing unit two (2) with reduced pressure, as well, a perimetritic frame 1. A wooden frame bottom 7 is made of 8 mm thickness plywood and there is fixed a heat sealing film to it 5 with foil. In the gap between a glazing unit and frame bottom, there is laid a thin walled flexible stainless corrugated tube 4∅16 mm in the coil form. Tube edges are fixed to the inlet and outlet protruding tubes 6.

Table 1: Technical Indicators of Flat Solar Collector

Parameters	Value
Absorbing plate material	Copper
Absorber plate dimensions	2 m×1 m
Plate thickness	0.4 mm
Glazing material	Hardenedglass
Glazing dimensions	2 m×1 m
Glazing thickness	4 mm
Insulation	Foam plexus (foam polyurethane)
Collector's tilt	45°
Absorber's thermal conductivity	401 W/(m K)

Insulation thermal conductivity	0.04 W/(m K)
Transmission-absorption factor	0.855
Visual sun temperature	4350 K
Atmospheric temperature	303 K
Irradiation intensity	1000^2

2.1 Substantiation of Design-Technological Scheme

Due to the necessity of upgrading the installation's operational specifications, we have developed a principle diagram of double circuit solar collector with a heat pump (Figure 3).

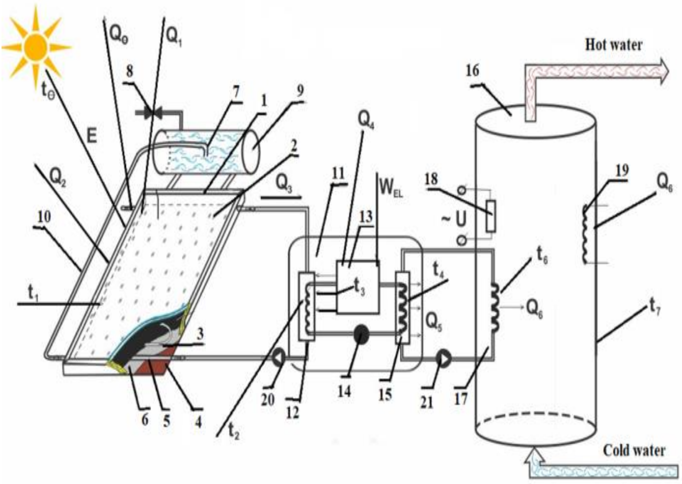


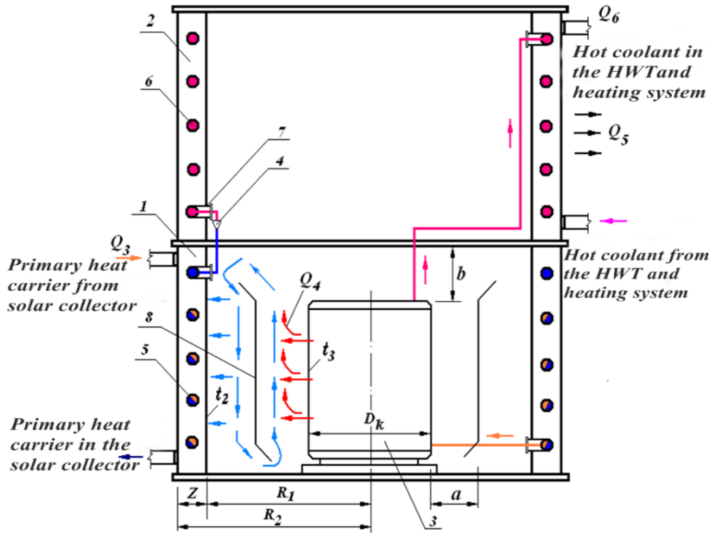
Figure 5: Functional Diagram of Double Circuit Solar Installation with Thermo Syphon Circulation.

The assumed installation's operation is fulfilled as follows:

Solar energy E with temperature t_0 is absorbed by the solar collector 1, with the temperature t_1, which, heating the solar energy flow, passes through translucent insulating double glass unit 2. The heat, obtained from the solar stream, heats the liquid in the coils 3, which is removed from the collector, and replaced with cold water from the water pipe with a tap for cold water 8, and in the dosing unit syphon 7 there takes place constant thermosyphon circulation by means of circulation tube 10. Further, the liquid flows into a heat pump 11, which consists of 12 **-** condenser with a temperature t_2, in which a heat exchanger is made in the coil form, absorbing the thermal medium heat, reduces its temperature lower than the atmospheric air temperature (Q_2) by means of a speed control valve 14, thereby helping the additional heat absorption from the atmospheric air. The scheme also shows the solar irradiation, reflected from semitransparent coating (Q_0) and absorbing panel surface (Q_1). In the heat pump, there is fulfilled a heat carrier power transfer with a relatively low temperature, to a condenser's heat exchanger transfer medium 15 in the coil form with higher temperature t_2, which increases the square, as well as the heat exchange intensity. To execute such a cycle, there is used a compressor 13 with temperature t_3, electric drive 17. Further, by means of the condenser heat exchanger 15 with temperature t_4, the heat from the heat pump (Q_5) is transferred to the heat exchanger's storage tank Q_6 with temperature t_6 of the heating system 18. As the installation has two circuits, it is provided with automatic circulation pumps 19 and 20 for liquid circulation between the solar collector and evaporator, condenser and storage tank. Water temperature is conditioned to the demanded technological level and provided the consumers with hot water and heating supply [15].

The main element of the studied circuit is the thermal pump. Figure 2 gives the diagram of the new technical solution for the thermal pump in section. The proposed device allows utilizing the heat, released by a compressor in the

operation process and simultaneously cool it.



1 – evaporator heat exchanger; 2 – condenser heat exchanger; 3 - compressor; 4 – throttling valves 5 and 6 – pipes for evaporator and condenser cooling medium; 7- hole for pipes entry into condenser's body; 8 – grilled barrel-divider of air flows.
Figure 6: Design Diagram of the Thermal Pump New Technical Solution for Solar Plant and Main Structural Parameters.

Table 2: Main Thermal Pump Structural Parameters Coolant

Outer diameters of evaporator and condenser (D_2), mm	375,0
Inner Diameter (D_1), mm	325 and 307
Height of heat exchanger jackets, mm	355 and382
Heat exchangers jackets width (Z), mm	20 and30
Heat exchangers jackets volumes, dm^3	0.52 and0,8
Core barrel-divider diameter, mm	238
Excess of evaporator height over compressor height, mm	80,0

Heat exchangers of evaporator 1 and condenser 2 are made in ring-type form vessels, formed with inner and outer cylindrical barrels with radii R_1 and R_2 and are installed one over another coaxially, underneath-evaporator, above-condenser, forming internal cylindrical air chamber. In the evaporator, heat exchanger chamber there is installed a compressor 3. To secure optimal heat exchange from a compressor to an evaporator, in the gap between them there is placed a cylindrical grilled barrel-divider 8. Inside the ring-type vessels, there circulated the transfer media of the 1st and 2nd circuits, removing the heat from pipes 5 and 6. Thus, a compressor practically is inside «a cold pot», the walls of which are cooled with the tubes cooling media 5. Consequently, the heat, released with a compressor is absorbed with a thermal pump evaporator, upgrading its productivity, and a compressor is simultaneously cooled without a ventilator usage.

Figure 7: Thermal Pump.

Analysis has shown, that the thermal pump's new solution for the solar heat supply system, without accounting the energy expenditures on the refrigerating ventilator drive promotes the heat productivity raise.

2.2 Methodology of Computing Electric Energy Consumption by Thermal Pump System and Seasonal Operation Performance includes the Following Stages

- forming the equations systems of the thermal pump heating balance
- drawing up and calculating the temperature field schemes on the surfaces of compressor, evaporator and condenser;
- forming the equations of convective heat exchange in the heat pump;
- forming the equations of distribution on the heat pump surface;
- thermal pump compressor capacity calculation;
- computing the optimization of the heat exchanger "tube in tube";
- thermal T- S diagram, Diagram of substance state, i-s-diagram.

The heat pump operates in a complex system associated with a low potential source, is involved in heat exchange with the external environment. Accordingly, all losses in the thermal pump can be broken down into two groups: external and internal. Figure 8 gives the classification of all losses, accompanying the operation process.

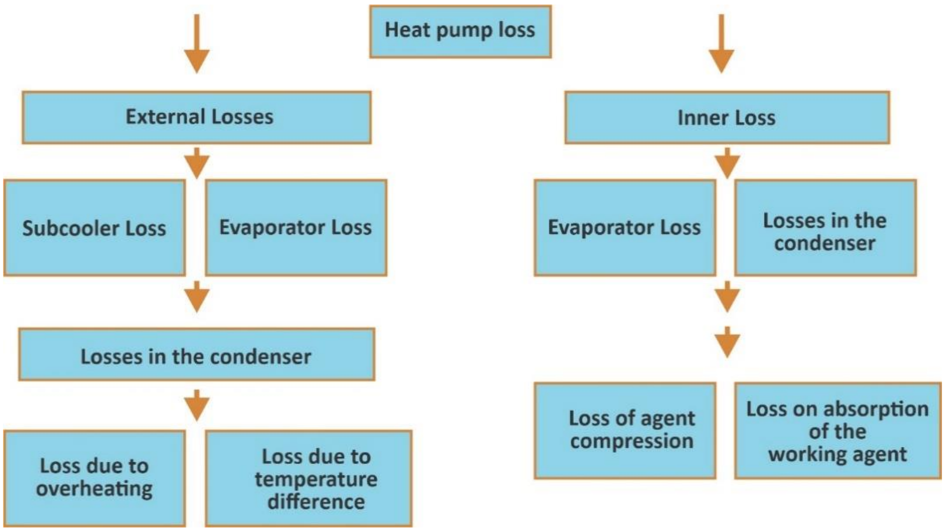


Figure 8: Classification of Losses in Thermal Pump.

2.1. Analysis of Thermal Pump Internal Losses

Internal losses are associated with the state processes of the working agent within the installation. External losses are isolated in the heat transfer processes occurring in a low-grade heat source. Values of internal losses are indicated with the type of equipment, the size of its main elements and working conditions. Thus, specific power consumption on transformation of heat in the real thermal pump will be considerably higher, than for an ideal process, as it is practically impossible to eliminate completely all losses in the thermal pump. By means of technical and economic assessment, we can obtain only some feasible losses decrease.

To assess the losses on thermal pump system elements, let's consider actual thermodynamic working process of real steam-compression single-step heat pump in T-S diagram (Figure 9).

Operation process consists of the following cycles:

a-2 –compression of working substance in the compressor;

2-x–steam cooling to saturation state (overheat removal);

x-3 –active agent condensation in the condenser;

2-3 –working agent cooling and condensation in the condenser;

3-4 –Liquid working agent cooling in super cooler;

4-5 –working agent throttling effect;

5-1 –working agent evaporation in evaporator;

1-a–working agent overheating prior to compression in the compressor.

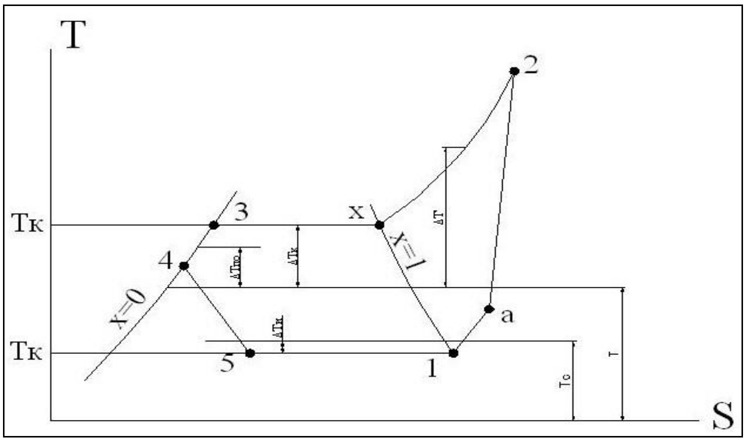


Figure 9: Operation Process of Real Steam-Compression Single Step Thermal Pump.

In the transition from the ideal to the real process, the sources of losses are sequentially introduced, the process of irreversibility of which leads to an increase in the mechanical work spent on the reverse cycle in a real thermal pimp:

- Carnot reverse cycle - the nature of thermodynamic superiority;
- the reverse Rankine cycle and the theoretical cycle of the heat pump have one internal source of irreversibility - losses due to throttling of the working substance (process 4-5 in Figure 9);
- heat pump with external and partially internal irreversibility
- external (due to the inaccessibility of the difference between temperature and condensation of the working agent, due to the finite difference in temperature and boiling of the working agent) and partially internal irreversible losses;
- a heat pump with a real pump was added due to the following sources of reversibility: internal irreversibility due to non-isentropic compression of the working agent in the compressor (process-2 in Figure 9), external irreversibility during heat release;
- the working agent in the capacitor (process 2 in Figure 9), external irreversibility due to the temperature difference between the capacitor and the transmission medium (3-4 in Figure 9).

However, this will increase the surface area of the heat exchangers and the consumption of mechanical energy when passing the working agent through such heat exchangers.

The degree of perfection of the cycle and, therefore, the degree of perfection of the heat pump substantially depend on the loss in compression of the active agent in the compressor and on the thermodynamic properties of the working agent. To build a cycle in a T-S diagram of a (P) diagram, you need to know the temperature at its specific points: working agent boiling temperature in evaporator $t_{и}$, working agent condensation temperature in the condenser$t_{к}$and the temperature in front of the control choke$t_{в}$. Important, as well, the working agent temperature value before the inlet into the compressor t_{pa}. Tentatively, the set temperatures are determined by means of simplified dependencies based on the experience of operating heat pumps. The boiling point is taken depending on the temperature of the potential heat source, the type and design of the heat exchanger. T is calculated by the formula

$$\Delta t_u = t_{H2} - \Delta t_u \quad (1)$$

where Δt_u – temperature drop in low potential sources $ Δt_u=2÷ 15°C; t_{H2} – temperature of heat low-temperature level at the outlet from evaporator.

Δt_u value might be computed, if to know the heat exchanger heat effectiveness. Condensation temperature is defined according to the formula:

$$t_к = t_{B2} - \Delta t_к \quad (2)$$

where: t_{B2} – temperature of heat consumer at compressor's outlet; $\Delta t_К$-"under heating" in a condenser. Since the consumer of heat is water, where the temperature is from 2 to 12 ° C. The calculation of its value is given below. The temperature of the liquid working agent in front of the control choke (Figure 9):

$$t_B = t_{B1} - \Delta t_{по} \quad (3)$$

where t_{B1} —heat consumer temperature at inlet to the heat pumpTH; $\Delta t_{по}$= 5-20°C-"under heating" in the super cooler.

Working agent temperature before in let to a compressor t_{pa} is defined with two factors: evaporation temperature t_0 and working agent overheating upon absorbing into compressor $\Delta t_{кр}$. Working agent over heating upon absorption excludes moisture ingress into the compressor, which is very dangerous for the piston compressor, as the liquid in "hazardous" space can cause an accident.

$$t_{pa} = t_1 + \Delta t_{кр} \quad (4)$$

where t_1 – working agent temperature at the outlet from evaporator.

$\Delta t_{кр}$ magnitude for single step compressors on ammonia equals to 5-10 °C, for freon's10-35 °C. Magnitudes t_u in the above given formulae (3,4) depend on the heat exchanger thermal performance. For the heat exchanger-heater it equals to:

$$E = (t''_{хт} - t'_{хт}) / (t'_{гт} - t'_{хт}) \quad (5)$$

For heat exchanger cooler

$$E = (t'_{гт} - t''_{гт}) / (t'_{гт} - t'_{хт}) \quad (6)$$

In the above formula, the index "gt" means the working body, the index "xt" - cold working body. One stroke marks the thread, two strokes mark the exit from the heat exchanger. Using the equations for calculating the heat exchanger, we obtain the formula for calculating the thermal characteristics of the heater:

$$E=\frac{W_{\Gamma T}}{W_{XT}}\cdot\frac{1-\exp\left(-\left(1-\frac{W_{\Gamma T}}{W_{XT}}\right)*\frac{\psi F_{\Gamma T}}{W_{\Gamma T}}\right)}{1-\frac{W_{\Gamma T}}{W_{XT}}\exp\left(-\left(1-\frac{W_{\Gamma T}}{W_{XT}}\right)*\frac{\psi F_{\Gamma T}}{W_{\Gamma T}}\right)} \quad (7)$$

where: $W_{\Gamma T}$ и W_{XT} –water equivalents of hot and cold working bodies; к–heat transmission factor; $F_{\Gamma T}$— surface from the side of hot working body; φ- coefficient, considering the flow path nature [1].

For contra flow φ= 1, for simple cross flow path φ= 0,6-0,7.

Water equivalents equal to product of expenditures by the working body heat capacity, i.e.

$$W_{\Gamma T} = G_{\Gamma} c_{pr} \quad (8)$$

$$W_{xT} = G_{x\Gamma} c_{px} \quad (9)$$

For heat exchanger cooler the ratio $W_{\Gamma T} / W_{xT}$, standing in front of fraction in (9), equals to a unity element. If water equivalents are equal, i. e. $W_{\Gamma T}$= W_{xT}= W, then from (9) it is easy to obtain the known equation[2]:

$$E = \left(1+\frac{W}{\psi F}\right) \quad (10)$$

Being aware of thermal performance (10), we can find the temperature of the body, being heated at the outlet from the heat exchanger.

$$t''_2=t'_2+E\,(t'_1=t'_2) \quad (11)$$

$$t_{B2}=t_{B1}+E\,(t_{\kappa}=t_{B1}) \quad (12)$$

Where t_{B1}–water temperature (heat consumer) at condenser's inlet.

From (12) it follows, that:

$$\Delta t_{\kappa}=t_{\kappa}+t_{B2} \quad (13)$$

Having substituted here the magnitude t_{B2} from (13), we will define:

$$\Delta t_{\kappa}=(t_{\kappa}+t_{B1})\cdot(1-E) \quad (14)$$

If $t_{\kappa}+t_{B1}$= 25 °C, then upon E =0,6 $t_{\kappa}+t_{B1}$ =10°C, and upon E = 0,85]Δt_{κ} = 3.75°C [3], which is close to the numbers, recommended above. Similarly, we might obtain the formulae for computation of Δt_{κ}, $\Delta t_{\pi o}$, $\Delta t_{\kappa p}$ dependent on thermal effectiveness of the corresponding heat exchanger.

Subsequent to specifying the cycle representative temperatures there is conducted defining the thermodynamic

parameters of the cycle points in the following sequence:

- p.1 –steam at saturation linex= 1 for boiling temperature$t_н$;
- p.a–over heat estimate a for critical pressure $P_к$and temperature

$$t_a = t_1 + \Delta t_2 \quad (15)$$

- p.3 –at saturation linex= 0 at the known temperaturet_K;
- p.2 — overheated steam for the pressure, which is Pк, at$S'_2 = S_a$
- p.4 –liquid phase for pressure, which is Pк;
- p.5 –wet steam for pressure, which is P_0, at $i'_4 = i_4$

The temperature of the liquid working body (agent), being cooled in the heat exchanger, is defined from its thermal balance. After constructing the cycle in the diagram there is executed the computation of thermal pump basic parameters:

Compressor's actual internal operation:

$$A_K = (i_2 - \frac{i_a}{\eta_K}) \quad (16)$$

Working body enthalpy at compressor's outlet

$$i_2 = i_a + A_K \quad (17)$$

Specific heat flow in evaporator:

$$q_0 = i_5 + i_1 \quad (18)$$

Specific heat flow in condenser:

$$q_K = i_2 + i_3 \quad (19)$$

Specific heat flow in super cooler:

$$q_{пsc} = i_3 + i_4 \quad (20)$$

Specific heat flow in the thermal pump:

$$q_{Thp} = i_к + i_{tp} \quad (21)$$

Working agent expenditures.

$$G = Q_B/(q_K + q) = Q_B/q, \text{ кг/с} \quad (22)$$

where Q_B–heat pump thermal capacity.

Volume flow at compressor inlet

$$V_0 = G * V_a \text{м}^3/\text{с} \tag{23}$$

where: V_a — working agent specific volume

Heat flow in evaporator:

$$Q_{ev} = G * q_u \text{кВт.} \tag{24}$$

Heat flow in super cooler:

$$Q_{sc} = G * q_{\text{по}} \text{кВт.} \tag{25}$$

Thermal pump transformation coefficient:

$$\mu = q / A_K \tag{26}$$

Energy expenditures:

$$Э_{\text{T}eex} = N_{hp} \text{А}_\text{к} \tag{27}$$

Compressor's electric capacity:

$$N_{cec} = Э_{hp} * Q_\text{К} \tag{28}$$

3. RESEARCH

The paper discusses the experimental data of the cooling agent R134a on the performance of the heat pump. As a result, data were obtained on the operation of the compressor, the coefficient of performance, the cooling capacity of the heat pump condenser, and the effect of the mass air flow in the evaporator jacket on the heat pump system was studied. The numerical results of the obtained simulation are presented in graphical form.

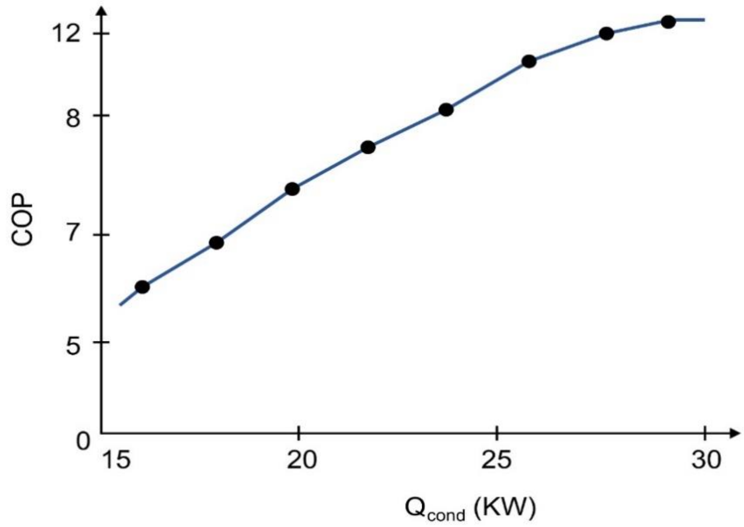


Figure 11: Dependence of Water Mass Flown Condenser Heat Capacity in the Thermal Pump.

Figure 11shows, that at the same water temperature at inlet there is increased the condenser heat capacity at the expense of water mass flow.

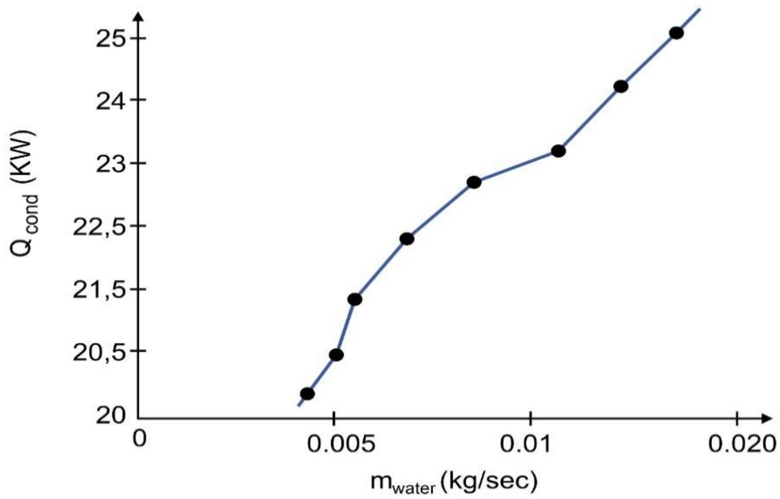


Figure 11: Dependence of the Thermal Pump Condenser Heat Capacity on Productivity Factor.

Figure 11 demonstrates the dependence of the thermal pump condenser heat capacity on productivity factor, cooling capacity has been constant. During the thermal pump research, it has been proved, that condenser's heat capacity, efficiency is increased.

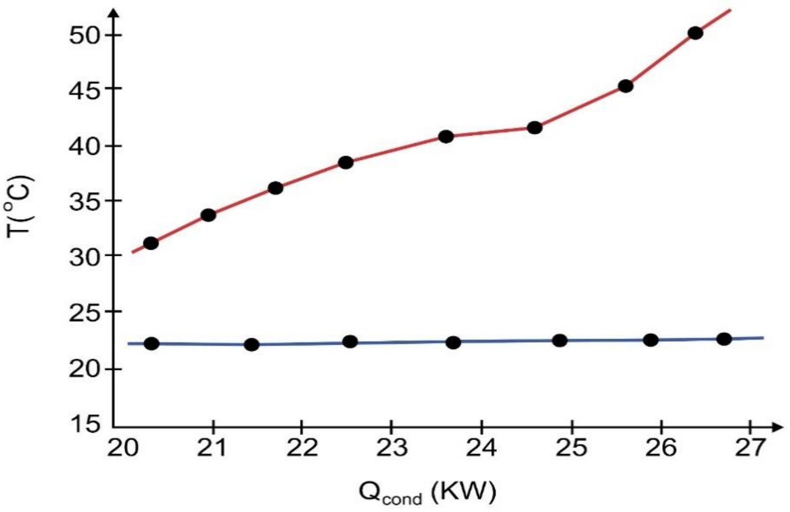


Figure 12: Dependence of Condenser Heat Capacity Change on the System Temperature.

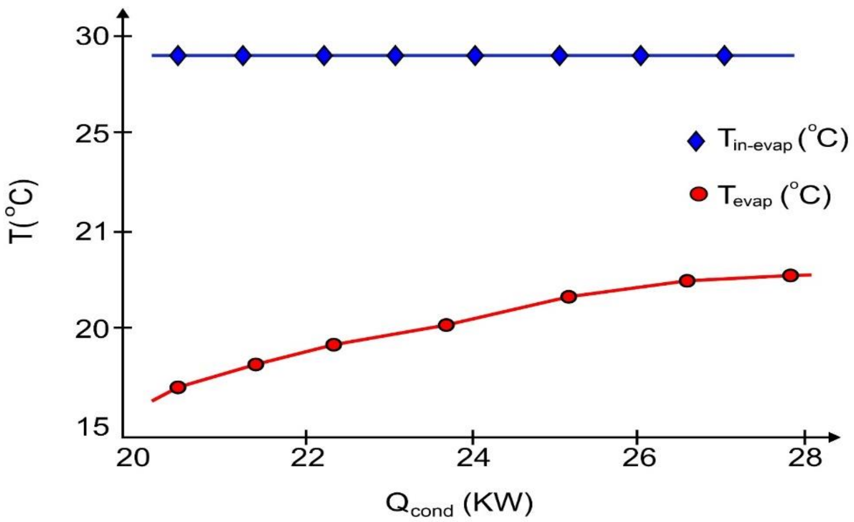


Figure 13: Dependence of Condenser Heat Capacity Change on the System Temperature.

Figures 12 and 13 show the condenser heat capacity change efficiency on the system temperature. Studied data do not depend on cooling agent temperature in the heat exchanger. Temperature gradient at condenser outlet is increased along with grow the of condenser heat capacity, but there is a little change of air temperature at evaporator outlet with variation for the condenser heat capacity.

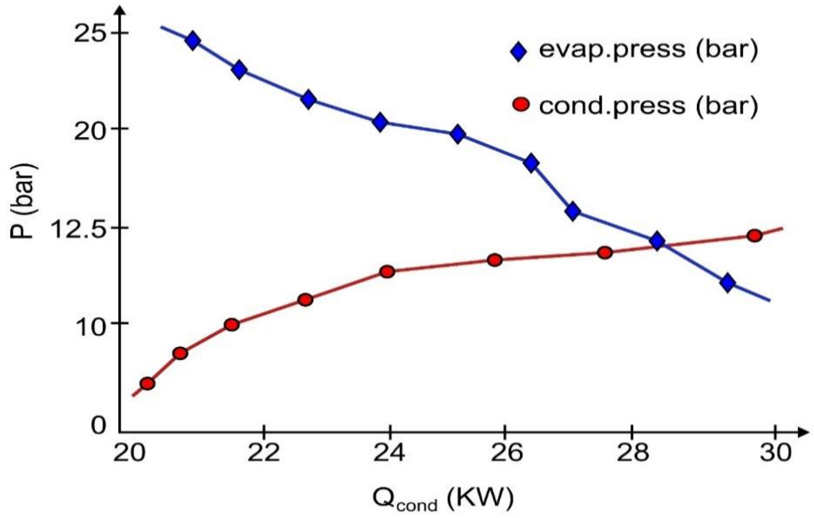


Figure 13: Dependence of Condenser Pressure and Evaporator Pressure on the Thermal Pump Heat Capacity.

Figure 13 shows the dependence of pressure in condenser and evaporator on the thermal pump heat capacity. As we see on the Figure, pressure in thermal pump condenser is decreased, while the evaporator pressure is increased along with the thermal pump condenser heat capacity increase.

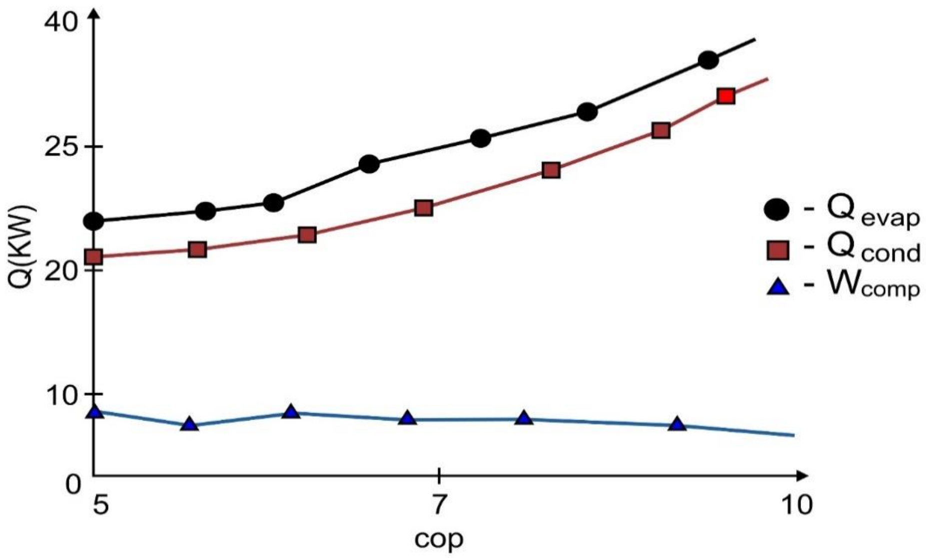


Figure 14: Dependence of Heat Capacity Change on Water Mass Flow.

Figure 14 shows dependence of heat capacity change on the water mass flow. As it is seen from that dependence, the pressure drop between the condenser and evaporator is increased upon the system cooling, compressor operation is increased due to the condenser heat capacity decrease.

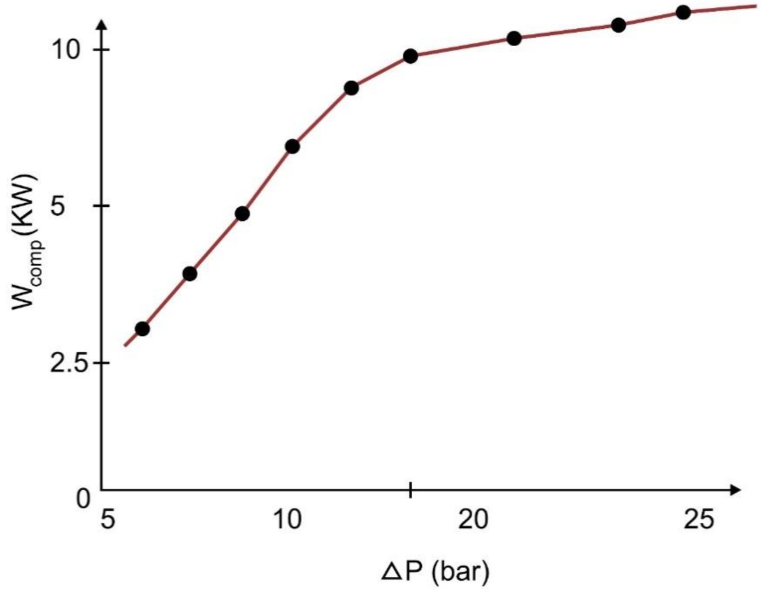


Figure 15: Dependence of Thermal Pump Energy on Pressure Change.

Figure 15 demonstrates the research dependence of thermal pump energy on pressure change. Form the given dependence, we can assume, that when the compressor operates, consumption is decreased, consequently, there is increased the efficiency. And when the condenser heat capacity grows, there is increased the efficiency, and cooling capacity is practically the same.

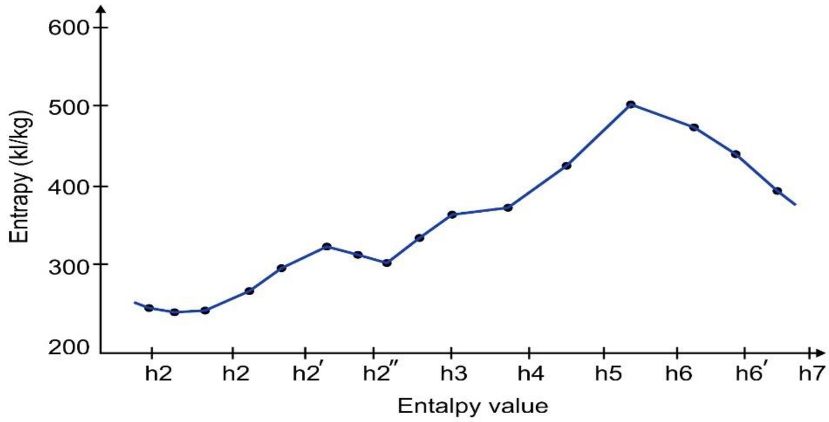


Figure 16: Dependence of Entropy Change on Enthalpy in Thermal Pump.

5. CONCLUSIONS

Experimental out comes for seasonal performance of air-water heating system within two temperature ranges reasonably confirm, that the thermal pump system effectively operates in the heating mode under accurately measured external air temperature, typical of climatic zone of Almaty city (Kazakhstan). Economic analysis shows, that expenditures on the consumed electric energy from the thermal pump system for winter heating season within the temperature intervals, being considered, are extremely effective. Research outcomes can be used both upon designing the thermal pumps systems of air-water type, and upon searching the energy efficient buildings.

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